

# **ANALYSIS OF HEAT REJECTION TO WATER BODIES**

**By  
R. C. SEHGAL**

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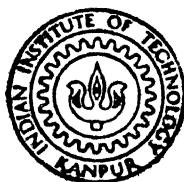
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**DEPARTMENT OF MECHANICAL ENGINEERING  
INDIAN INSTITUTE OF TECHNOLOGY, KANPUR  
SEPTEMBER, 1978**

# **ANALYSIS OF HEAT REJECTION TO WATER BODIES**

**A Thesis Submitted  
in Partial Fulfilment of the Requirements  
for the Degree of  
MASTER OF TECHNOLOGY**

**By  
R. C. SEHGAL**

**to the  
DEPARTMENT OF MECHANICAL ENGINEERING  
INDIAN INSTITUTE OF TECHNOLOGY, KANPUR  
SEPTEMBER, 1978**

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## CERTIFICATE

This is to certify that the thesis entitled  
'Analysis of Heat Rejection to Water Bodies' by  
Mr. R.C. Sehgal is a record of the work carried out  
under my supervision and has not been submitted else-  
where for a degree.

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POST GRADUATE OFFICE

This thesis has been submitted  
for the award of the degree of  
Master of Technology  
in the subject of  
Mechanical Engineering  
and is certified to be  
of original research  
and contains no material  
published elsewhere.

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(R. C. SEHGAL)

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## NOMENCLATURE

A	Surface area of the water body
c	Cloudiness factor
$c_p$	Specific heat of fluid at constant pressure
e	Partial vapour pressure
$e_s$	Saturated vapour pressure
F	Body force
h	Heat transfer coefficient
H	Depth of the water body
i, j	Index numbers for grid points
k	Thermal conductivity
L	Transverse length of water body
$L_1$	Distance of the outfall opening from the x - axis
$L_2$	Distance of the intake opening from the y - axis
l	Width of intake and outfall openings
M	Total number of finite divisions along y - axis
N	Total number of finite divisions along x - axis
n	Unit normal vector
p	Hydrostatic pressure
Q	Rate of net heat loss per unit surface area from the water body
$Q_a$	Long wave radiation from the atmosphere
$Q_{br}$	Back radiation from the water body
$Q_c$	Convective heat loss
$Q_e$	Evaporative heat loss

$Q_i$	Incident solar radiation
$Q_p$	Rate of heat rejected by the power plant per unit area of the water body
$Q_r$	Reflected radiation
$Q_s$	Effective solar radiation
$Q_w$	Long wave radiation from water body
Re	Reynolds number, ( $Re = \frac{U_1 l}{\nu}$ )
R.H.	Relative humidity
r	A counter for the number of iterations
t	Temperature at a point in the water body
T	Dimensionless temperature, ( $T = \frac{t - t_e}{\Delta t}$ )
$\Delta t$	Difference between the intake and outfall temperatures
$t_a$	Temperature of the ambient medium
$t_e$	Equilibrium temperature
$t_s$	Surface temperature of the water body
U	Uniform velocity of fluid at intake and outfall
V	Velocity of fluid at any point in the water body
$V_w$	Wind velocity
u, v	Velocity components in the x, y directions
$\frac{u}{U}, \frac{v}{U}$	Dimensionless velocity components in the X, Y directions
W	Length of the water body
x, y	Cartesian co-ordinates

X, Y Dimensionless Cartesian co-ordinates

$\Delta x$  Length of the finite element

### Greek Symbols

$\beta$  Sky radiation factor

$\rho$  Specific density

$\nu$  Eddy viscosity

$\nu_H$  Eddy diffusivity

$\sigma$  Stefan Boltzman Constant

$\psi$  Stream function

$\bar{\psi}$  Dimensionless stream function

## ABSTRACT

In this study, the governing differential equations and the corresponding finite difference equations, for one dimensional and two dimensional horizontal flow models, have been developed to determine numerically the steady state velocity field and the resulting temperature distribution in the water body, which receives the heat rejected from a thermal source and dissipates this energy to the environment. The flow pattern for the two limiting cases of inviscid and creeping flow, as well as for flow at finite low Reynolds numbers, have been analysed. The effect of various parameters, such as the aspect ratio of the water body, intake and outfall locations, flow rate, eddy diffusivity etc., have been investigated. The study provides guidelines for limiting the power plant capacity for a given water body and determines the effect of the power plant capacity on the intake temperature rise.

## CHAPTER I

### INTRODUCTION

#### 1.1 The Problem

The importance of heat rejection to water bodies is primarily due to its use in thermal and nuclear power plants, where large quantities of cooling water are utilised to condense the exhaust steam from the turbine as water. For an efficient and satisfactory operation of these plants, it is essential to have an efficient system for heat rejection.

The continued growth of electric power supply is vital for our industry and, thus, for our national economy. It is imperative that this growth be accomplished economically and with an environmental impact which is on acceptable level. As the growth of industry based on electric power continues, unit sizes, plant capacity and site locations present new challenges. The bulk of the electric power, at present, is being generated through the use of steam driven thermal power plants and these plants use large quantities of cooling water to condense the exhaust steam from the turbine. This keeps the temperature of condensation low so as to maximize the energy conversion. The condensate is recirculated back to the boiler and the cooling water is generally heated by 8 to 12 °C, through energy exchange

in the condensers. One of the most important factors to be considered in selecting plant locations is the method of waste heat rejection and the availability of water resources to accomplish it.

Cooling water is generally obtained from natural, or artificial, lakes or from rivers, and is returned to the source at a higher temperature. Ultimately, almost all this energy from the cooling water, is dissipated to the atmosphere through energy exchange at the surface, with a negligible loss to the bed of the lake or the river. This rejection of heat from the cooling water alters the naturally existing temperature balance in a body of water.

The heat transfer mechanisms which occur at the surface of a cooling water body comprise of the following:

- (a) incoming short-wave solar radiation,
- (b) incoming long-wave atmospheric radiation,
- (c) portions of both short-wave and long-wave radiation which are reflected by the water surface,
- (d) long-wave back radiation from the water to the atmosphere,
- (e) heat exchange due to conduction and convection,
- (f) heat loss due to evaporation and gain due to condensation.

The combined effect of all these heat transfer mechanisms determines the rate of cooling for water bodies under specific conditions. The last three of the above heat transfer mechanisms, i.e., back radiation, conduction-convection and evaporation are functions of the water body surface temperature. This water temperature is subject to control through variations in plant design or operating practices. Although the quantity of heat to be rejected is relatively fixed for a certain plant capacity, condenser design and water flow rate can be manipulated to achieve different values of discharge temperature and flow velocity.

The maximum temperature increase at any given point and the affected area or volume can be estimated for specific conditions. The maximum temperature can be depressed by mixing the returning cooling water as thoroughly as possible with ambient water. At the other extreme, the cooling water may be floated over the ambient water as a stratified layer, maintaining high surface temperature. The latter scheme has the advantage over the former in that the rate of heat loss to the atmosphere is maximum, and, therefore, the area affected is minimum, even though local temperatures are higher. At the same time, the intake for the power plant has to be located on a lake or river, so as to obtain water at the lowest possible temperature for a high thermal efficiency and a discharge located to prevent warm water recirculation to the intake.

## 1.2 Previous Work

Not much work has so far been done in this area. Initially, this area was considered mainly due to the thermal pollution effects arising from heat rejection. The need of a fluid flow and heat transfer analysis has been realised only during the last few years, with a view to improve the power plant efficiency.

Steady laminar flow, with closed streamlines at large Reynolds number, was analyzed by Batchelor [1]. The effects of various factors on the amount of recirculation of water, after it has been used as a coolant in the thermo-electric plants and returned to the river, or lake, from where it was originally drawn, was perhaps first carried out by Bata [2]. Some of the more important elements which need be considered in a well designed circulating water system were highlighted by Richards [3].

Raphael [4] presented a procedure for predicting the temperature of various water bodies, such as shallow lakes, flowing streams and detention reservoirs, from weather records, the inflow and outflow characteristics, the surface area and the volume of the water body. This procedure assumes that the thermocline is absent and the water so stirred by wind or internal currents that the temperature is uniform throughout. A mathematical model

for predicting temperatures in rivers and reservoirs was developed by Delay and Seaders [5]. They computed the net monthly energy exchange from the reservoir and distributed it vertically so that the resulting temperature gradients approximated the standard for each month. Dake and Harleman [7] formulated a theoretical model for the time dependent vertical temperature distribution in a deep lake during the yearly cycle by using heat-flux balance at the water surface, which accounts for back radiation and evaporative loss, as a boundary condition.

It was realised later on that to determine the methods for minimizing the recirculation of waste heat, in the design of cooling water systems for thermal power plants, it was desirable to understand the nature of mixing of the warm water discharge. Accordingly several experiments on the mixing of the buoyant jet of water being discharged horizontally at the surface of a body of water were performed in the laboratory by Yuan, Wiegel and Ismail [6] and by Tamai, Wiegel and Tornberg [8]. The results obtained from these experiments compared favourably with the actual data obtained from different thermal power plants. Subsequently it was shown by Stefan [10, 13] that the flow and cooling of a warm water surface layer could be represented in a physical

model if, first, flow of water in and out of the model, and second, atmospheric conditions, it was exposed to, were controllable. He simulated the phenomena of mixing at the outlet, stratification, and the surface heat transfer through the use of physical models. An experimental design to simulate a heated water discharge from a channel into a deep lake for lateral spread was also developed.

Brown [9] discussed the various methods for the disposal of waste heat and concluded that studies were needed to develop improved and less costly cooling device that would be required increasingly at new plant locations. He also emphasised the need to explore the possibilities of a beneficial use of the waste heat. The alternative methods of heat dissipation from power plants and the economic evaluation of these alternatives was presented by Hauser Loius [11]. Analysis for reducing the cooling pond area and water loss, by designing the pond for higher operating temperatures, was presented by Brue and Alden [12].

The thermal effects of power plants on lakes in terms of its temperature cycle were studied by Moore and Jaluria [14, 15]. A one dimensional model based on the assumption of a constant Richardson number at the base of the stratified layer was developed. The model gave a

constant heat flux across the thermocline. The model was then perturbed in terms of heat flux as well as the thermal diffusivity to give the power plant impact on a lake. Based upon the assumption of horizontal isotherms at all times, Huber, Harleman and Ryan [16] developed a mathematical model to predict the vertical temperature distribution in stratified reservoirs. The model included the effects of distribution of heat within the reservoir, by convection and diffusion heat sources and sinks at boundaries, and internal dissipation of solar radiation. Temperature profiles predicted from the solution of this mathematical model agreed well with measured values in both the field and the laboratory. Hindley and Miner [17] observed that the temperature distribution in a cooling water body, being employed for a power plant, was dependent upon the manner in which the heated effluent mixed with the receiving water body and upon the rate of heat exchange between the water surface and the atmosphere. It was found that, far from the receiving end, where mixing was nearly complete, the dominant cooling mechanism was heat exchange with the atmosphere. The rate of this cooling was directly proportional to the surface heat exchange coefficient, which in turn is a function of several environmental variables, particularly the wind velocity.

Grubert and Abbott [18] formulated a mathematical model for nearly horizontal stratified flows. They showed that the characteristic directions in a stratified fluid divide into pairs, each pair being associated with a fluid layer. Miyazaki [19] studied the heated two dimensional jet discharged at the water surface and carried out an analysis at various Richardson numbers for determining the velocity and temperature distributions.

Jaluria, Variyar and Mehta [20, 21] formulated a mathematical model to predict the temperature and velocity profiles in a body of water, due to the basic mechanisms in the presence of wind, cloudiness, back radiation etc. They computed the equilibrium average surface temperature, which is defined as the temperature attained by the surface, if the ambient conditions are held constant at specific values; and studied the heat transfer mechanisms at the surface in detail. They also developed a more generalised model, considering convection effects caused by the outflow and inflow of cooling water and the heat addition from the power plant. Certain specific, simplified, one and two dimensional models were studied by them in detail.

### 1.3 Present Work

In the present work, one dimensional and two dimensional horizontal flow models have been considered. Corresponding finite difference equations have been

developed to determine numerically the steady state velocity field and the resulting temperature distribution in the water body, which receives the heat rejected from a thermal source and dissipates this energy to the environment. These models are related mainly to the problem of heat rejection from a power plant to a water body, such as a lake, river, sea etc. These may also represent the problems of heat rejection from the condensers of an airconditioning plant to a water pond or the heat rejection from industrial systems to water bodies.

A nonlinear fourth order partial differential equation, in the form of nondimensionalized variables, is obtained for the flow, in a two dimensional model for the water body, neglecting variations in the vertical direction. The finite difference formulation of this equation in the explicit form was used for computing the recirculation at different Reynolds numbers. The recirculation was also computed for the two limiting cases, of inviscid and creeping flow. The effect of various parameters, such as aspect ratio of the water body, the intake and outfall locations and eddy viscosity, on the nature of recirculation was studied. The resulting velocity field was obtained and the general nature of flow discussed in terms of streamlines and velocity distributions.

For finding the temperature distribution, a one-dimensional model, neglecting variations in the vertical and lateral directions, was initially studied. The effect of heat rejection from the power plant on the temperature distribution in the water body was computed and its dependence on various parameters, such as length of the water body, temperature difference between intake and outfall temperatures, flow rate etc., have been investigated. The effect of turbulence has been considered in terms of the eddy diffusivity  $v_H$ . The net surface heat exchange was computed by using a method similar to that outlined by Jaluria, Variyar and Mehta [20], with respect to ambient conditions such as solar flux, wind speed, ambient temperature, cloudiness factor, etc.

In the second phase of the present work, a two-dimensional model, neglecting variations in the vertical direction, was developed to determine the temperature distribution in the water body. The temperature distribution has been computed for creeping flow and for flows at small values of the Reynolds number. The impact of the intake and outfall locations on recirculation was studied so as to obtain a system with minimum intake temperature rise, for rejecting a particular quantity of thermal energy. The effect of various parameters, such as dimensions of the water body, intake and outfall temperature

difference, flow rate, etc., on the temperature distribution has been considered in detail. The physical significance of these results with respect to actual systems is discussed.

All the computations were carried out on IBM 7044 at I.I.T., Kanpur. The meteorological data needed for the study has been obtained largely from Indian Meteorological Department and partly from the analysis given by Raphael [4].

## CHAPTER 2

### FORMULATION OF THEORETICAL MODELS AND METHOD OF SOLUTION

The behaviour of a water body, which acts as a cooling water source for a power plant, is determined, to a large extent, by the nature of flow field set up due to the outfall of warm water from the condensors of the power plant and by its interaction with the environment, to which it ultimately dissipates the heat rejected by the power plant. Hence, in developing an analytical model for a cooling water body, the mathematical representation of the various mechanisms underlying its energy exchange with the environment is of considerable importance.

This chapter considers the analysis and the methods of solution for different cases of fluid flow in the water body, under steady state conditions. The basic features and assumptions made in carrying out the analysis are outlined as follows:

- (a) Finite dimensions of the water body.
- (b) Energy transfer takes place at the intake and outfall of the cooling water for the power plant and at the surface of the water body. Energy transfer from the sides and the bottom of the water body is neglected. This assumption is adequately supported by various investigations.

- (c) An idealized configuration, of vertical sides and flat horizontal base and top surface, is considered. This can eventually be modified to consider, as close as possible, the exact topology of a given water body.
- (d) Though ambient conditions are functions of time, they are assumed to have specific values over a considerable period of time.
- (e) The effect of viscous dissipation is neglected.
- (f) The physical properties of the fluid are assumed constant, with respect to temperature variation.

Any other approximations employed in addition to the above, have been mentioned in the course of analysis.

## 2.1 Recirculation Models

The amount of recirculation, to the cold water intake, of the heated water directly affects the efficiency of operation of a power plant and also usually lowers its maximum capacity. It depends on the geometry of the intake, the design of the outfall, the distance between the intake and outfall, the discharge and mean depth of the flow in the water body, the relative amount of diversion and the degree of heating of the diverted water.

The analysis proceeds by applying the principle of conservation of mass, momentum and energy to a differential control volume at any point in the water body. The basic equations, for incompressible flow under steady state conditions, are:

$$\nabla \cdot \bar{V} = 0 \quad (2.1)$$

$$(\bar{V} \cdot \nabla) \bar{V} = \frac{1}{\rho} (\bar{F} - \nabla p) + v \nabla^2 \bar{V} \quad (2.2)$$

$$(\bar{V} \cdot \nabla) t = v_H \nabla^2 t \quad (2.3)$$

where  $\bar{V}$  denotes the velocity,  $\bar{F}$  the body force,  $t$  the temperature,  $v$  the eddy viscosity and  $v_H$  the eddy diffusivity. The net amount of heat exchange per unit area with the environment 'Q' appears as a boundary condition at the surface.

The solution of the above equations gives the velocity field and the temperature distribution in the water body. The equations are quite complex and are usually coupled through the temperature  $t$  because of the buoyancy force which arises from the  $(\bar{F} - \nabla p)$  term and is a function of  $t$ . Thus, these equations need to be further simplified before a solution is attempted.

A two dimensional model with complete mixing in the vertical direction is considered. This situation frequently arises in the case of a shallow water body.

The intake and outfall are assumed to be located at the same level, since due to vertical mixing a vertical separation can not be considered. The vertical direction being fully mixed, the buoyancy effect of the temperature variation with depth does not arise. As a result, the problem becomes a decoupled one, i.e., the flow pattern may be determined without considering the temperature field. The temperature variation in the horizontal plane may then be determined from the known flow field. The basic equations become:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (2.4)$$

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = - \frac{1}{\rho} \frac{\partial p}{\partial x} + v_x \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \quad (2.5)$$

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = - \frac{1}{\rho} \frac{\partial p}{\partial y} + v_y \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) \quad (2.6)$$

$$u \frac{\partial t}{\partial x} + v \frac{\partial t}{\partial y} = v_H \left( \frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} \right) + \frac{Q}{\rho c_p H} \quad (2.7)$$

where  $Q$  is the net amount of heat exchange per unit surface area and  $H$  the depth of the water body.

### 2.1.1 Governing Equations

The eddy viscosity at any point in the flow region for turbulent mixing in the directions  $x$  and  $y$  is assumed to be equal so that

$$v_x = v_y = v$$

The equation (2.4) can be satisfied identically by using a stream function given by

$$u = \frac{\partial \psi}{\partial y} \quad (2.8)$$

$$v = -\frac{\partial \psi}{\partial x} \quad (2.9)$$

The values of  $u$  and  $v$  in terms of the stream function are substituted in Equations (2.5) and (2.6). Then to eliminate the pressure term  $p$ , equation (2.5) is differentiated with respect to  $y$  and equation (2.6) with respect to  $x$  and the resulting equations are subtracted from each other to yield the following equation.

$$\frac{\partial \psi}{\partial y} \cdot \frac{\partial}{\partial x} (\nabla^2 \psi) - \frac{\partial \psi}{\partial x} \cdot \frac{\partial}{\partial y} (\nabla^2 \psi) = v (\nabla^4 \psi) \quad (2.10)$$

where

$$\nabla^2 \psi = \frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} \quad (2.11)$$

$$\text{and } \nabla^4 \psi = \frac{\partial^4 \psi}{\partial x^4} + 2 \frac{\partial^4 \psi}{\partial x^2 \partial y^2} + \frac{\partial^4 \psi}{\partial y^4} \quad (2.12)$$

The equation (2.10) is non-dimensionalized by using the following dimensionless parameters:

$$X = \frac{x}{l}, \quad Y = \frac{y}{l} \quad \text{and} \quad \bar{\psi} = \frac{\psi}{U l}$$

where  $U$  is the uniform velocity at inlet to the water

body and  $l$  is the opening at the inlet point. Here  $\bar{\psi}$  represents the dimensionless stream function. However for convenience of notation, this has been shown on Figures through  $\psi$ .

The non-dimensionalized equation thus obtained is

$$\frac{\partial \bar{\psi}}{\partial Y} \cdot \frac{\partial}{\partial X} (\nabla^2 \bar{\psi}) - \frac{\partial \bar{\psi}}{\partial X} \cdot \frac{\partial}{\partial Y} (\nabla^2 \bar{\psi}) = \frac{1}{Re} \nabla^4 \bar{\psi} \quad (2.13)$$

$$\text{where } Re = \text{Reynolds number} = \frac{U l}{v} \quad (2.14)$$

The solution of Equation (2.13) alongwith the requisite boundary conditions gives the velocity field for viscous flow. The equation is a nonlinear, partial differential equation and its general solution is not known. It is amenable to a numerical solution in finite difference form, being somewhat simpler at low Reynolds number. The solution can be obtained for three different cases outlined below.

#### (a) Inviscid Flow

The stream function for two dimensional inviscid flow, being irrotational, satisfies the Laplace equation i.e.,

$$\nabla^2 \bar{\psi} = 0 \quad (2.15)$$

with the following boundary conditions, as shown in Fig. 1.

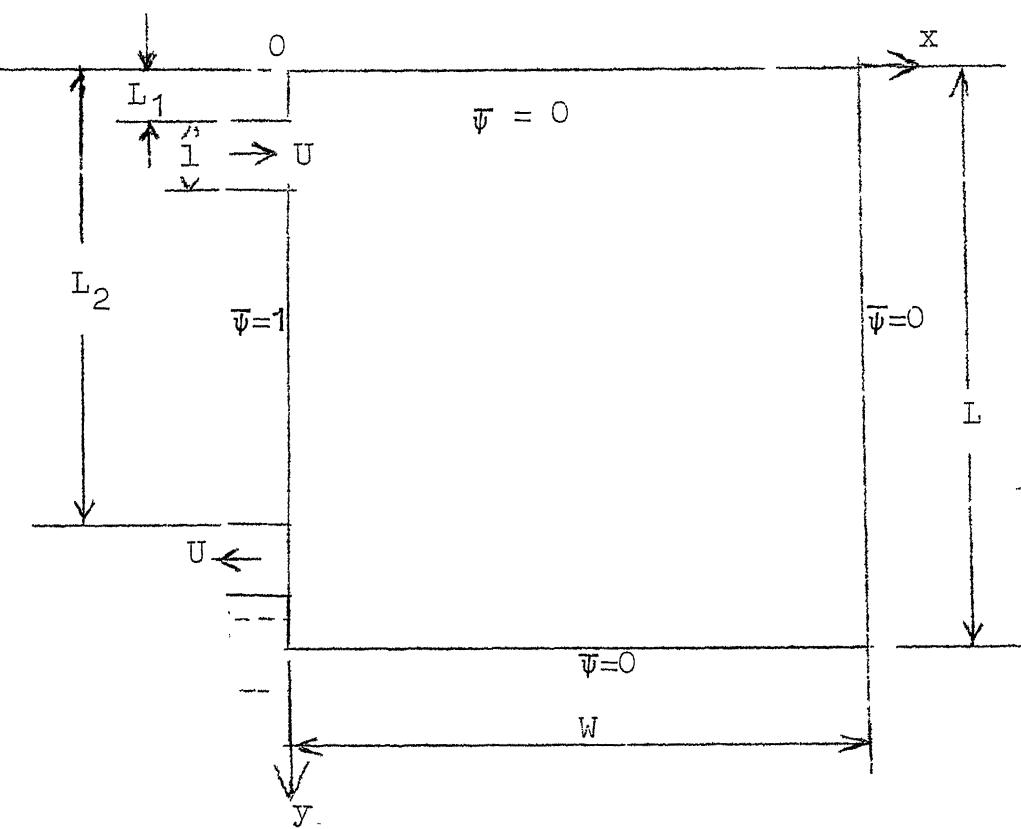


FIG. 1 TWO DIMENSIONAL HORIZONTAL FLOW MODEL  
SHOWING INTAKE AND OUTFALL LOCATIONS

$$x = 0, 0 \leq y < L_1; \quad \frac{\partial \bar{\psi}}{\partial \bar{y}} = 0$$

$$x = 0, L_1 \leq y \leq (L_1 + 1); \quad \frac{\partial \bar{\psi}}{\partial \bar{y}} = U; \quad \frac{\partial \bar{\psi}}{\partial \bar{Y}} = 1$$

$$x = 0, (L_1 + 1) < y < L_2; \quad \frac{\partial \bar{\psi}}{\partial \bar{Y}} = 0$$

$$x = 0, L_2 \leq y \leq (L_2 + 1); \quad \frac{\partial \bar{\psi}}{\partial \bar{y}} = -U; \quad \frac{\partial \bar{\psi}}{\partial \bar{Y}} = -1$$

$$x = 0, (L_2 + 1) < y \leq L; \quad \frac{\partial \bar{\psi}}{\partial \bar{Y}} = 0$$

$$x = W, 0 \leq \bar{y} \leq L; \quad ; \quad \frac{\partial \bar{\psi}}{\partial \bar{Y}} = 0$$

$$y = 0, 0 \leq x \leq W; \quad ; \quad \frac{\partial \bar{\psi}}{\partial \bar{X}} = 0$$

$$\text{and } y = L, 0 \leq x \leq W; \quad ; \quad \frac{\partial \bar{\psi}}{\partial \bar{X}} = 0$$

This implies constant  $\psi$  value at the walls as shown in Fig. 1.

(b) Creeping Flow:  $Re \ll 1$

If the Reynolds number is very small, the R.H.S. of Equation (2.13) dominates the flow and the inertia terms on the L.H.S. can be neglected. This is the special case of creeping flow, where viscous effects predominate and inertia terms are negligible. The equation then reduces to

$$\nabla^4 \bar{\Psi} = 0 \quad (2.16)$$

This is a fourth order linear equation known as the biharmonic equation in two dimensions. The solution of this equation alongwith the following boundary conditions gives the velocity field for creeping flow.

$$\begin{aligned}
 & x = 0, \quad 0 \leq y \leq L_1 \quad ; \quad \frac{\partial \bar{\Psi}}{\partial Y} = \frac{\partial \bar{\Psi}}{\partial X} = 0 \\
 & x = 0, \quad L_1 < y < (L_1 + l); \quad \frac{\partial \bar{\Psi}}{\partial Y} = 1 \text{ and } \frac{\partial \bar{\Psi}}{\partial X} = 0 \\
 & x = 0, \quad (L_1 + l) \leq y \leq L_2 ; \quad \frac{\partial \bar{\Psi}}{\partial X} = \frac{\partial \bar{\Psi}}{\partial Y} = 0 \quad (2.16a) \\
 & x = 0, \quad L_2 < y < (L_2 + l) ; \quad \frac{\partial \bar{\Psi}}{\partial X} = 0 \text{ and } \frac{\partial \bar{\Psi}}{\partial Y} = -1 \\
 & x = 0, \quad (L_2 + l) \leq y \leq L ; \quad \frac{\partial \bar{\Psi}}{\partial X} = \frac{\partial \bar{\Psi}}{\partial Y} = 0 \\
 & x = W, \quad 0 \leq y \leq L ; \quad \frac{\partial \bar{\Psi}}{\partial X} = \frac{\partial \bar{\Psi}}{\partial Y} = 0 \\
 & y = 0, \quad 0 \leq x \leq W ; \quad \frac{\partial \bar{\Psi}}{\partial X} = \frac{\partial \bar{\Psi}}{\partial Y} = 0
 \end{aligned}$$

$$\text{and } v = L, \quad 0 \leq x \leq W; \quad \frac{\partial \bar{\psi}}{\partial X} = \frac{\partial \bar{\psi}}{\partial Y} = 0$$

This implies constant value of  $\psi$  at the walls and also the value of the gradient as given above.

### (c) Flow at Higher Reynolds Numbers

For obtaining the velocity field at different Reynolds numbers, Equation (2.13), with all the terms, is solved, alongwith the boundary conditions given by equation (2.16 a). Even for very large Reynolds numbers, the R.H.S of Equation (2.13) can never be negligible near a solid boundary because the no slip condition forces  $\nabla^4 \bar{\psi}$  to be very large, of order  $Re$ , near the wall.

#### 2.1.2 Method of Solution

Equations (2.13), (2.15) and (2.16) are solved by using finite difference techniques. Employing a rectangular mesh, the surface area of the water body is divided into M parts along the y - axis and N parts along the x - axis, each part being of length  $\Delta x$ . The various partial derivatives are replaced with their central difference approximations, which convert the governing differential equations into a set of simultaneous algebraic equations. Appendix I gives the finite difference form of the equations. The finite difference equations are solved by using the Gauss - Siedel iterative method.

Before starting the iteration scheme, all the mesh points need to be initialized. The computer time needed for achieving a given criterion for convergence depends on this initialization and as such the guess for this should be as close to the exact solution as possible.

The rate of convergence for inviscid flow is much faster than for other flows due to the lower order of the governing equation. Therefore, the Equation (2.15) is solved first by using the following initialization.

$$\bar{\psi}_{i,j}^0 = 0 \quad \text{for all } (i, j)$$

where  $(i, j)$  represents the interior mesh points in the flow area.

The solution obtained for inviscid flow is used as an initial guess for solving the equation (2.16). Finally the creeping flow solution is used as an initial guess for solving Equation (2.13). The solution of Equation (2.13) for higher Reynold numbers is obtained by using the value of stream function for lower values of Reynolds numbers, close to the new values, as an initial guess.

Starting from these initial values of  $\bar{\psi}_{i,j}^0$ ; the successive iterates  $\bar{\psi}_{i,j}^1, \bar{\psi}_{i,j}^2, \dots, \bar{\psi}_{i,j}^r$ , are obtained by using the finite difference equations

given in Appendix I. The iteration is continued till convergence is achieved i.e., the difference between  $\bar{\psi}_{i,j}^{r+1}$  and  $\bar{\psi}_{i,j}^r$  becomes smaller than a specified value for all values of  $i, j$ , where  $r$  is a given iteration.

## 2.2 Energy Exchange at the Surface

The energy exchange at the surface is due to various heat transfer mechanisms outlined below:

### (a) Radiation from the Sun and from the Atmosphere

At a point outside the atmosphere of the earth, radiant energy is received from the sun, on a surface normal to its rays, at a rate of approximately  $1165 \text{ K}_\text{c} \text{al}/\text{hr Sq. m}$  [4]. This value is called the 'Solar Constant'. A plane normal to the rays of the sun at the surface of the earth receives considerably less energy as compared to the solar constant because moisture, gases and solid particles in the air scatter and absorb much of the incident radiation. Although much of the radiant energy is lost, some of this energy is recovered as diffuse radiation.

Of the total quantity of the direct, or short wave, radiation that reaches the surface of a water body, a portion is returned unchanged due to reflection at the surface and scattering by bubbles and suspended particles

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immediately below it [4]. However, for practical engineering computations, solar radiation and reflected radiation can be combined in one function, ( $Q_i - Q_r$ ) or :

Effective absorbed solar radiation =  $Q_s = Q_i - Q_r$

If  $c$  is the average cloud cover in tenths of sky covered, the net short wave radiation is obtained from the correlation given by Raphael [4]

$$Q_s = (1 - 0.0071 c^2) (Q_i - Q_r) \quad (2.17)$$

#### (b) Back Radiation

Effective back radiation may be defined as the difference between  $Q_w$ , the longwave radiation leaving a body of water, and  $Q_a$ , the longwave radiation from the atmosphere being absorbed by the body of water [4]. Atmospheric radiation does not follow any simple law as it is a function of many variables, such as the distribution of moisture, temperature, ozone, carbon dioxide, etc. Thus, this has been found to be dependent on cloud height, as high, middle and low clouds, and the amount of cloud, as scattered cloud, broken cloud and overcast. However, as most weather observations give cloud amount in tenths of sky obscured, the variation of the atmospheric radiation factor has been given as a function of cloud amount and vapour pressure by Raphael [4]. The effective back

radiation,  $Q_{br}$  can be computed as,

$$Q_{br} = Q_w - Q_a = 0.970 \times \sigma (T_s^4 - \beta T_a^4) \quad (2.18)$$

where  $\sigma$  is Stefan-Boltzman constant,  $T_s$  is absolute temperature of water surface,  $T_a$  is absolute ambient temperature and  $\beta$  is sky radiation factor, tabulated by Raphael [4].

### (c) Convection

Sensible heat is convected to or from a body of water whenever a temperature difference exists between air and water. The basic equation for the convection heat transfer,  $Q_c$ , from the water surface to the air is,

$$Q_c = h (T_s - T_a) \quad (2.19)$$

where  $h$  is the surface heat transfer coefficient. The value of  $h$  is largely dependent on the condition of the air in contact with the water and hence it is a function of the wind velocity, its temperature, direction and turbulence level etc. The following correlation by Wada [23] has been found to be most suitable for conditions relevant to India.

$$h = 2.77 \times 10^{-3} (0.48 + 0.272 V_w) \text{ K Cal/m}^2 \text{ sec } {}^\circ\text{C} \quad (2.20)$$

where  $V_w$  is the wind velocity along the water surface in meter/sec.

## (d) Evaporation

When the vapour pressure of ambient air is less than the saturated vapour pressure, at the water surface temperature, water evaporates into the air, removing heat from the water mainly due to the energy required to evaporate the water and to a small extent by the secondary heat transfer due to the sensible heat contained in the water removed by evaporation. The heat transfer by evaporation is given by Hindley and Miner [17]

$$Q_e = 2 h [e(T_s) - e(T_a)] \quad (2.21)$$

where  $e(T_s)$  is the partial vapour pressure at water surface temperature in mm of Hg,  $e(T_a)$  is the partial vapour pressure at ambient temperature in mm of Hg and  $h$  is the heat transfer coefficient given by equation (2.20).

Also

$$e(T_a) = e_s(T_a) \times R.H.$$

where  $e_s(T_a)$  is the saturated vapour pressure at ambient temperature and R.H. is the relative humidity at ambient temperature.

The rate of net heat loss,  $Q$ , from the surface is given by

$$Q = Q_e + Q_{br} - Q_s + Q_c \quad (2.22)$$

When heat is being rejected by a power plant to a water body, the average surface temperature rises and the equilibrium condition is obtained when

$$Q_p = Q_e + Q_{br} - Q_s + Q_c \quad \text{or} \quad Q = Q_p \quad (2.23)$$

where  $Q_p$  is the rate of heat rejected by the power plant per unit area of the water body surface.

## 2.3 One Dimensional Model

Physically, a one dimensional model represents the circumstance where mixing in the water body in two directions is so vigorous that exchange of heat and mass need be considered only in the third direction. One dimensional horizontal model with energy exchange at the surface and variation in the longitudinal direction, is considered. Therefore, temperature in the vertical and lateral directions is assumed to be uniform. The theoretical model, under steady state conditions, is developed below.

### 2.3.1 Governing Equations

The surface of the water body is assumed to consist of a number of finite divisions, each of width  $\Delta x$ . The finite difference equations are obtained by considering the energy balance for each element i.e.,

$$\text{Energy input} - \text{Energy output} = \text{Losses.}$$

## (d) Evaporation

When the vapour pressure of ambient air is less than the saturated vapour pressure, at the water surface temperature, water evaporates into the air, removing heat from the water mainly due to the energy required to evaporate the water and to a small extent by the secondary heat transfer due to the sensible heat contained in the water removed by evaporation. The heat transfer by evaporation is given by Hindley and Miner [17]

$$Q_e = 2h [e(T_s) - e(T_a)] \quad (2.21)$$

where  $e(T_s)$  is the partial vapour pressure at water surface temperature in mm of Hg,  $e(T_a)$  is the partial vapour pressure at ambient temperature in mm of Hg and  $h$  is the heat transfer coefficient given by equation (2.20).

Also

$$e(T_a) = e_s(T_a) \times R.H.$$

where  $e_s(T_a)$  is the saturated vapour pressure at ambient temperature and R.H. is the relative humidity at ambient temperature.

The rate of net heat loss,  $Q$ , from the surface is given by

$$Q = Q_e + Q_{br} - Q_s + Q_c \quad (2.22)$$

When heat is being rejected by a power plant to a water body, the average surface temperature rises and the equilibrium condition is obtained when

$$Q_P = Q_e + Q_{br} - Q_s + Q_c \quad \text{or} \quad Q = Q_P \quad (2.23)$$

where  $Q_P$  is the rate of heat rejected by the power plant per unit area of the water body surface.

## 2.3 One Dimensional Model

Physically, a one dimensional model represents the circumstance where mixing in the water body in two directions is so vigorous that exchange of heat and mass need be considered only in the third direction. One dimensional horizontal model with energy exchange at the surface and variation in the longitudinal direction, is considered. Therefore, temperature in the vertical and lateral directions is assumed to be uniform. The theoretical model, under steady state conditions, is developed below.

### 2.3.1 Governing Equations

The surface of the water body is assumed to consist of a number of finite divisions, each of width  $\Delta x$ . The finite difference equations are obtained by considering the energy balance for each element i.e.,

$$\text{Energy input} - \text{Energy output} = \text{Losses.}$$

The governing equations for different elements have been developed in Appendix II. At intake and outfall, energy balance for half the elements is considered. The heat transfer due to conduction at intake and outfall has been assumed to be negligible. Thus three different equations have been developed, viz., one for the element at intake, one for the element at outfall and the third for all the intermediate elements.

These equations are solved alongwith the following boundary conditions.

- (a) Value of equilibrium temperature (i.e., temperature without heat rejection) throughout the water body is known.
- (b) Difference between the intake and outfall temperatures and the amount of heat to be rejected is known. This depends on the capacity of the thermal power plant and its efficiency.

The solution is obtained by using the iterative method explained in section 2.1.2.

#### 2.4 Two Dimensional Model

A two dimensional model represents the circumstance where mixing in one direction is so vigorous that energy exchange need be considered only in the

other two directions. A two dimensional model with heat transfer and flow in the longitudinal and lateral directions is considered. Thus a uniform temperature in the vertical direction is assumed, with temperature variations in the longitudinal and lateral directions only. A theoretical model, under steady state conditions, is developed below.

#### 2.4.1 Governing Equations

The surface of the water body is divided into a number of square meshes each of length  $\Delta x$ . The governing equations are obtained by considering the energy balance for each element as has been discussed in the case of the one dimensional model. Appendix III outlines these equations.

The equations have been solved for the temperature field throughout the water body by using the iterative method. The velocity field obtained from the corresponding governing equations, given in Section 2.1, has been used in these equations. The following boundary conditions have been used.

- (a) Same as for one dimensional flow.
- (b) Same as for one dimensional flow.
- (c) The boundaries of the water body are assumed to be adiabatic i.e.,  $\frac{\partial t}{\partial n} = 0$  where  $n$  is a unit normal vector at the boundary walls.

The energy exchange between the water body and the environment was calculated after obtaining the temperature field in the water body under steady state conditions. This energy was found to approximate to the heat rejection from the power plant.

## CHAPTER 3

### RESULTS AND DISCUSSION

This chapter presents the results obtained from the numerical solution of various governing equations developed in Chapter 2. The physical implications of these results have also been discussed. Of particular interest are the flow field, presented mainly in terms of stream function, and the temperature field, presented as isotherms.

The two dimensional, horizontal flow patterns in the water body, under steady state conditions, has been obtained from the numerical solution of Equations (2.13), (2.15) and (2.16). Figure 2 shows the streamlines in a water body, with intake and outfall located on the same side, for the two limiting cases of inviscid and creeping flow as well as for flows at two other Reynolds number values. The flow patterns for the inviscid and the creeping flow circumstances, with intake and outfall located on the ends of same side, are found to be symmetrical with respect to the centre line of the water body, whereas that for other Reynolds numbers is unsymmetrical due to the nonlinearity of the governing equations. A greater portion of the cooling water body is affected by the flow in the case of inviscid flow as compared to other flows.

Figure 3 shows the effect of aspect ratio of the water body on the flow field for creeping flow. It is found that the portion of the water body, which is away from the outfall of the hot water, remains largely undisturbed. Therefore, in this case, this portion of the cooling water body does not take part significantly in the heat rejection process. It is found that the flow spreads out to only about half the distance between intake and outfall. Figure 4 shows the flow pattern for the creeping flow circumstance in a water body whose longitudinal dimension is much smaller than the distance between the intake and outfall locations. It is found that almost the entire water body gets disturbed in this case and, thus, helps in the rejection of more heat per unit area. The effect of locating the intake and outfall openings nonsymmetrically, on the flow pattern in a water body, is shown in Figure 5. The flow pattern becomes unsymmetrical even for the creeping flow. The portion of the water body, which does not fall between the intake and outfall locations, is found to remain largely undisturbed and thus does not help in the removal of heat.

The results obtained indicate circulatory, motion near the corners of the water body due to the formation of eddy currents. The strength of these circulatory motions is, however, found to be very weak and,

therefore, the same have not been depicted on the Figures.

Figure 6 shows the variation of velocity components  $u$  and  $v$  along the X - axis and Figure 7 shows their variations along the Y - axis, both for creeping flow, with intake and outfall located on the same side of the water body. Figure 6 shows that the velocities have their maximum values near the outfall of the hot water and then decrease sharply away from it. The velocities drop to almost 10% of the maximum value even before half the length is reached. Therefore, the portion of the water body near the outfall is very significant in the removal of the rejected heat. Figure 7 shows that the velocity component  $u$  decays to zero along the axis, midway between intake and outfall locations and then starts picking up the corresponding values symmetrically but in the opposite direction.

The flow pattern in the water body with intake and outflow on the opposite sides of the water body is shown in Figure 8. Here also the inviscid flow affects a greater portion of the water body as compared to other cases. In this configuration, most of the water body is used. The effect of the aspect ratio of the water body on the stream function distribution for creeping flow is shown in Figure 9. It is found that the portion of the

water body away from the outfall of the hot water is also disturbed and this results in more effective heat rejection. Figure 10 shows another configuration of intake and outfall locations. It shows that the major portion of the water body remains largely undisturbed, if the intake and outfall are located on the centre of opposite sides. This is, therefore, a very ineffective method of heat rejection to the water body.

Here, the results for the flow field, for flows with Reynolds number below 28.0, only, could be obtained. When higher values of the Reynolds number were used, the finite difference equations became unstable. For calculating flow fields at higher Reynolds numbers, other methods such as the ADI method, employing a different finite difference scheme must be employed. In many of these methods, the transient problem is solved to reach the steady state solution.

The numerical solution of the energy equations, to obtain the temperature field, is considered next. The net energy lost by the surface of the water body,  $Q$ , as a function of the surface temperature, wind velocity, cloudiness, relative humidity, etc., is computed by a numerical evaluation of the various components of energy exchange outlined in Section 2.2. The energy exchange has been analysed for a particular set of meteorological parameters.

Figure 11 shows the effect of power plant capacity, i.e., of the heat rejected per unit surface area of the water body, and the effect of eddy diffusivity on the intake temperature obtained from a one dimensional horizontal flow model. The intake temperature increases with increase in  $Q$  as expected. The average surface equilibrium temperature,  $t_e$ , which is defined as the surface temperature at which the net heat exchange from the surface is zero, is given by the intercept of the curve on the temperature axis. It is found that the rate of increase in the intake temperature is small for lower values of  $Q$ . For higher values of  $Q$ , the intake temperature varies almost linearly. It is also found that the curve becomes slightly steeper with an increase in the value of eddy diffusivity. A higher value of eddy diffusivity results in greater energy transfer to the neighbouring fluid and would cause a larger rise in the intake temperature. Figure 12 shows the temperature decay in the direction of flow for various values of  $Q$ . As the hot water flows from the outfall to the intake, its temperature decreases due to the energy lost to the environment. For low values of  $Q$ , the temperature attains the equilibrium temperature before the flow reaches the intake, leaving the intake temperature unaltered. As the power level is increased, the increase in power being due to an increase in outfall temperature, the temperature throughout

the water body including the intake temperature increases. Figure 13 shows the effect of eddy diffusivity, or of turbulent mixing in the water body, on the temperature profiles. The intake temperature increases with an increase in the value of eddy diffusivity, as the increased value of eddy diffusivity implies more mixing in the water body which increases the rate of heat transfer across the water body. Thus a large amount of thermal energy is transferred to the adjacent layers and which, therefore, reflects a higher temperature at intake.

The isotherms in a two dimensional, horizontal, creeping flow are shown in Figure 14. The temperature decays away from the outfall as expected. A temperature rise of about  $3^{\circ}\text{C}$  at the intake is observed. The effect of outfall velocity on the temperature field is shown in Figure 15. The temperature difference,  $\Delta t$ , between outfall and intake is taken as constant at  $10^{\circ}\text{C}$ . As the power plant capacity, reflected in terms of higher velocity, increases, the temperature at the outfall of the hot water also increases. Therefore, the isotherms shift away from the outfall in the water body for this circumstance. It is found, however, that the isotherms are more spread out as the power plant capacity increases. Thus the surface area of the water body significantly involved in the rejection of heat increases. Figure 16 shows the

effect of power plant capacity, i.e., of the heat rejected, on the intake temperature. The rate of increase in the intake temperature, with respect to  $Q$ , is low for small values of  $Q$ , as was found in the one dimensional flow. But for higher values of  $Q$ , the intake temperature varies almost linearly and at a higher rate. The effect of Reynolds number on the isotherm distribution is shown in Figure 17. The isotherms have been shown with dimensionless temperature,  $T$ , which is defined as

$$T = \frac{t - t_e}{\Delta t}$$

It is found that there is a slight shift of isotherms inwards indicating a higher rate of heat rejection with increased value of Reynolds number. Thus the intake temperature decreases with an increase in Reynolds number, for the same amount of heat to be rejected. Reynolds number may increase because of an increase in the outfall velocity, which must also have a lower  $\Delta t$  for given heat rejected and would give the trends observed, or a decrease in eddy diffusivity, which was seen to give a similar effect earlier.

The results discussed in this chapter are predominantly for creeping flows as most of the flows we are concerned with in practice, for heat rejection to water

bodies, have low velocities when averaged for the two dimensional model. The actual problem is three dimensional. But to study shallow water bodies or the effect of horizontal separation of intake and outfall, this model may be employed.

## CHAPTER 4

### CONCLUSIONS

From the discussions presented in Chapter 3, the results obtained are found to be physically reasonable. A flow configuration of considerable practical significance is the one in which the intake and outfall are located on the same side, with a horizontal separation. The flow pattern obtained for this configuration indicates the limitations posed on the effective use of the longitudinal dimension of the water body, for heat rejection to the environment, mainly by the distance between the intake and the outfall. The flow field for the configuration with intake and outfall on the opposite sides of the water body have also been studied and these show that the entire surface area of the water body can be used effectively, in this case, for heat rejection and, thus, resulting in a lower rise in the intake temperature.

The effect of heat rejection from a power plant is found to be a rise in the intake temperature, which shows a greater change for a high power plant capacity, as expected. The effect of an increase in the value of the eddy diffusivity results, as expected, in an increase in the value of intake temperature because the increased value of eddy diffusivity implies more turbulent mixing,

which leads to an increased rate of heat transfer across the water body. The intake temperature is found to decrease marginally with an increase in Reynolds number over the range of low values of Re studied. This is expected, since Re increases due to an increase in the velocity or in the outfall width or due to a decrease in the eddy diffusivity.

The results obtained provide guidelines for limiting the power plant capacity for a given water body, with a particular horizontal distance available. The results are directly applicable to a shallow water body. Further, the results have been obtained for an idealized configuration, of vertical sides and flat horizontal base and top surface. To consider the arbitrary shape of the water body, the 1% streamline may be obtained and a simplified rectangular geometry can then be taken in the proximity of this streamline. To account for an actual shape is neither economical nor advisable. In order to obtain the flow and temperature fields for a general three dimensional problem, which is usually the case in practice, it is reasonable to solve for the two dimensional vertical and the horizontal flow patterns, which will give results to adequate accuracy. The vertical flow pattern will indicate the depth that may be employed for the horizontal

model and the flow spread of the latter gives the transverse dimension for averaging in the former model. These results may also be employed in vertical one dimensional models for studying the transient and stratification effects.

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## APPENDIX I

## FINITE DIFFERENCE EQUATIONS

The finite difference equations for two dimensional recirculation model are given below.

The equation (2.15) is represented in the finite difference form by

$$\bar{\psi}_{i,j} = \frac{(\bar{\psi}_{i+1,j} + \bar{\psi}_{i-1,j} + \bar{\psi}_{i,j+1} + \bar{\psi}_{i,j-1})}{4.0}$$

The equation (2.16) in finite difference form becomes

$$\bar{\psi}_{i,j} = \frac{8A - B - C}{20.0}$$

where

$$A = (\bar{\psi}_{i,j+1} + \bar{\psi}_{i,j-1} + \bar{\psi}_{i+1,j} + \bar{\psi}_{i-1,j})$$

$$B = (\bar{\psi}_{i+1,j+1} + \bar{\psi}_{i+1,j-1} + \bar{\psi}_{i-1,j+1} + \bar{\psi}_{i-1,j-1})$$

$$\text{and } C = (\bar{\psi}_{i+2,j} + \bar{\psi}_{i-2,j} + \bar{\psi}_{i,j+2} + \bar{\psi}_{i,j-2})$$

The finite difference form of equation (2.13) is given by

$$\bar{\psi}_{i,j} = \frac{8A - B - C}{20} + (AA \cdot DD - BB \cdot EE) (CC)$$

where

$$AA = (\bar{\psi}_{i+1,j} - \bar{\psi}_{i-1,j})$$

$$BB = (\bar{\psi}_{i,j+1} - \bar{\psi}_{i,j-1})$$

$$CC = \frac{Re}{80}$$

$$DD = (\bar{\psi}_{i+1,j+1} + \bar{\psi}_{i-1,j+1} - \bar{\psi}_{i+1,j-1} - \bar{\psi}_{i-1,j-1} \\ + \bar{\psi}_{i,j+2} - \bar{\psi}_{i,j-2} \text{ - } 4 \text{ BB})$$

$$\text{and } EE = (\bar{\psi}_{i+1,j+1} + \bar{\psi}_{i+1,j-1} - \bar{\psi}_{i-1,j+1} - \bar{\psi}_{i-1,j-1} \\ + \bar{\psi}_{i+2,j} - \bar{\psi}_{i-2,j} \text{ - } 4 \text{ AA})$$

## APPENDIX II

## ONE DIMENSIONAL FLOW

One dimensional flow, with temperature variation in the longitudinal direction only, is considered.

The surface area of the water body is divided into  $N$  number of finite elements, each of length  $\Delta x$  and width  $L$ . The governing equations are obtained by considering the energy balance for each element i.e.,

$$\text{Energy Input} - \text{Energy Output} = \text{Losses}$$

The solution of energy balance equation gives

$$t_i = \frac{t_{i-1} + t_{i+1}}{2.0} + DD(t_{i-1} - t_{i+1}) - \frac{Q}{Q_c}$$

where

$$Q_c = \frac{2 \rho C_p H v_H}{\Delta x^2}$$

$$DD = \frac{U}{4} \frac{\Delta x}{v_H}$$

and  $t_i$  represents the temperature at any point in the flow field except at intake and outfall.

For finding the temperature at intake and outfall, the energy balance is considered for half the element i.e., with length  $\frac{\Delta x}{2}$ , and the conduction term at intake and outfall is taken to be zero. Thus for each element, we get an algebraic equation as the governing equation.

## APPENDIX III

## TWO DIMENSIONAL FLOW

Two dimensional horizontal flow, with negligible temperature variation in the vertical plane, is considered.

The surface area of the water body is divided into M parts, each of length  $\Delta x$ , along the y axis and N parts, each of length  $\Delta x$  again, along the x - axis. The general energy equation for each element with two dimensional flow is given by

$$u \frac{\partial t}{\partial x} + v \frac{\partial t}{\partial y} = \frac{k}{\rho C_p} \left( \frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} \right) + \frac{Q}{\rho C_p H}$$

where Q denotes the energy transfer per unit area.

The conduction term at intake and outfall are taken to be zero, in the above equation, while finding the intake and outfall temperatures. The above equation is written in the finite difference form for each element so as to give as many number of equations as are the number of elements. These equations are then solved in the same manner as was done for one dimensional flow.

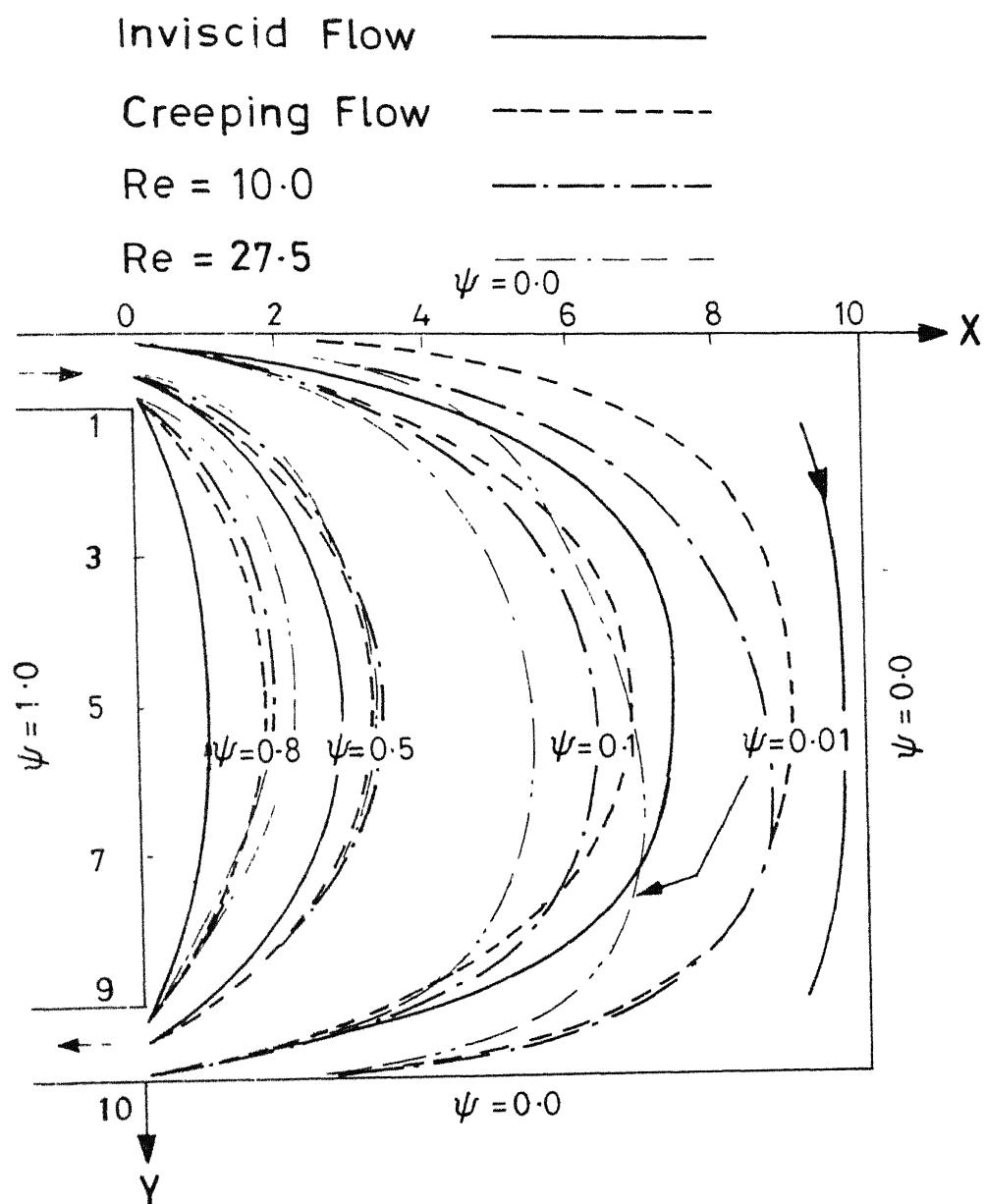


FIG. 2 FLOW PATTERN IN TWO DIMENSIONAL HORIZONTAL FLOW FOR VARIOUS REYNOLDS NUMBERS, WITH INTAKE AND OUTFALL ON THE SAME SIDE OF THE WATER BODY

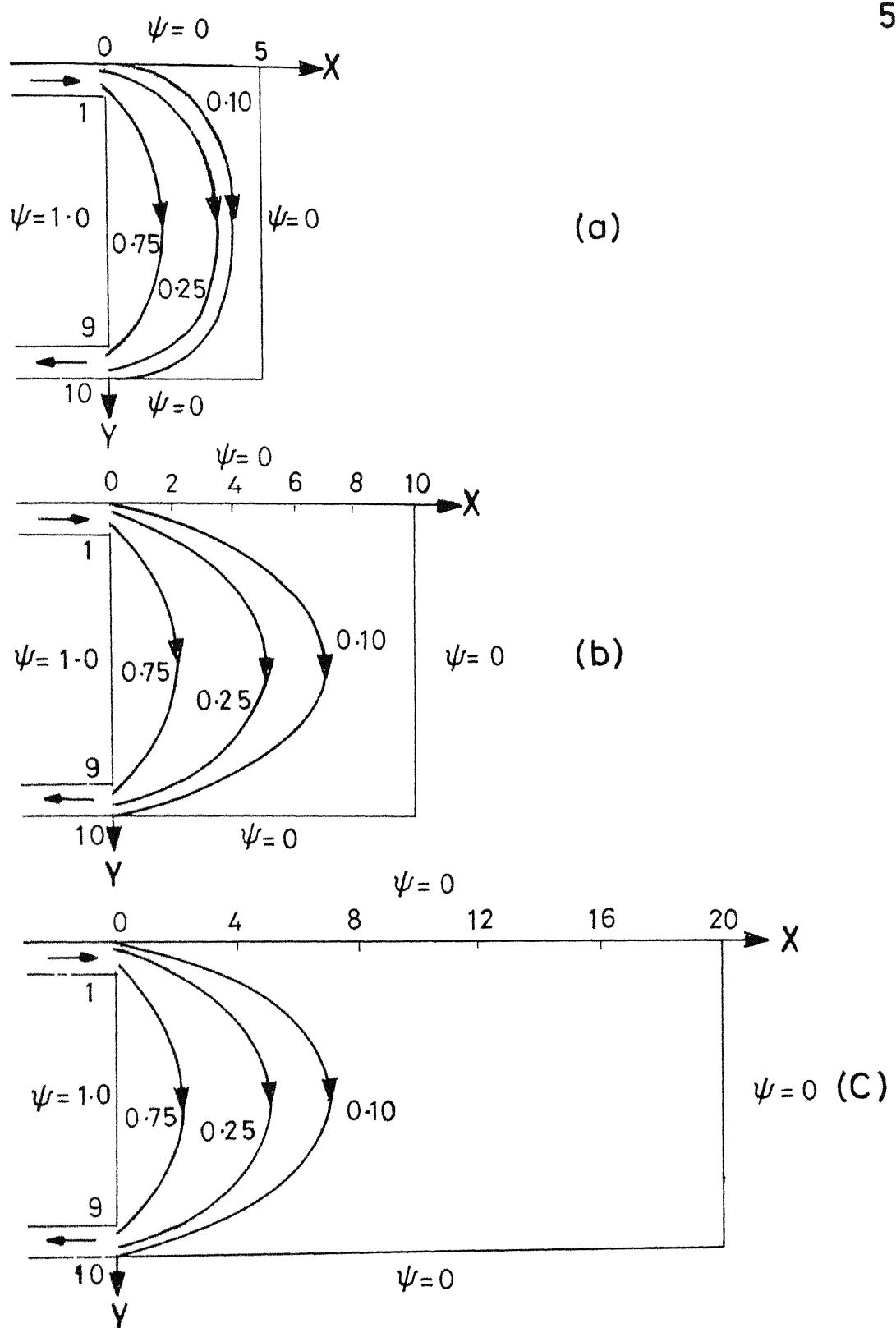


FIG. 3 FLOW PATTERN IN THE TWO DIMENSIONAL HORIZONTAL CREEPING FLOW, WITH INTAKE AND OUTFALL ON THE SAME SIDE FOR VARIOUS ASPECT RATIO'S OF THE WATER BODY

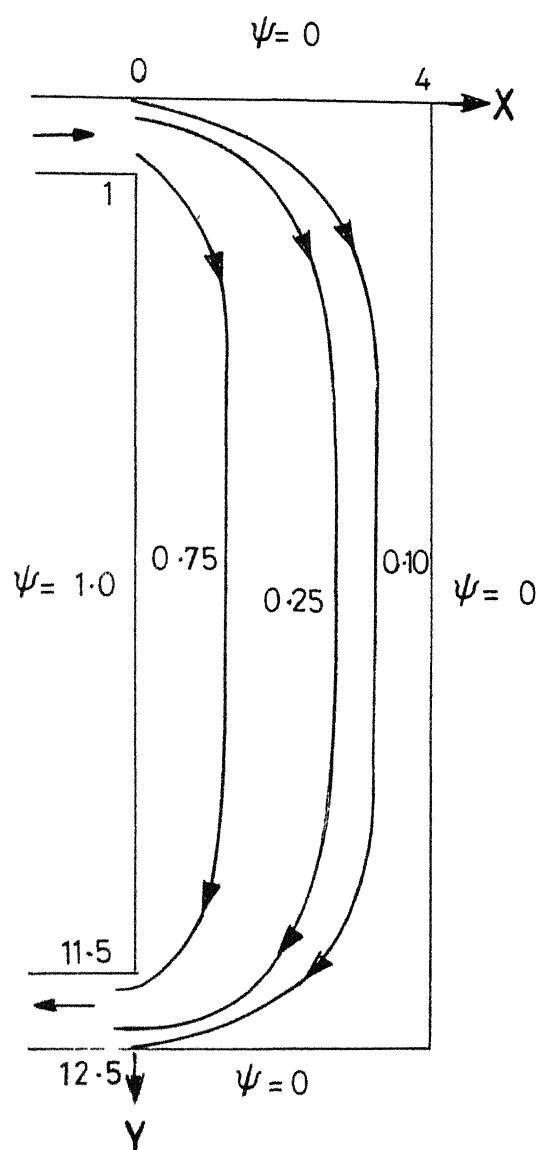


FIG. 4 EFFECT OF REDUCING THE LONGITUDINAL DIMENSION OF THE WATER BODY ON THE FLOW PATTERN IN THE TWO DIMENSIONAL HORIZONTAL CREEPING FLOW, WITH INTAKE AND OUTFALL ON THE SAME SIDE

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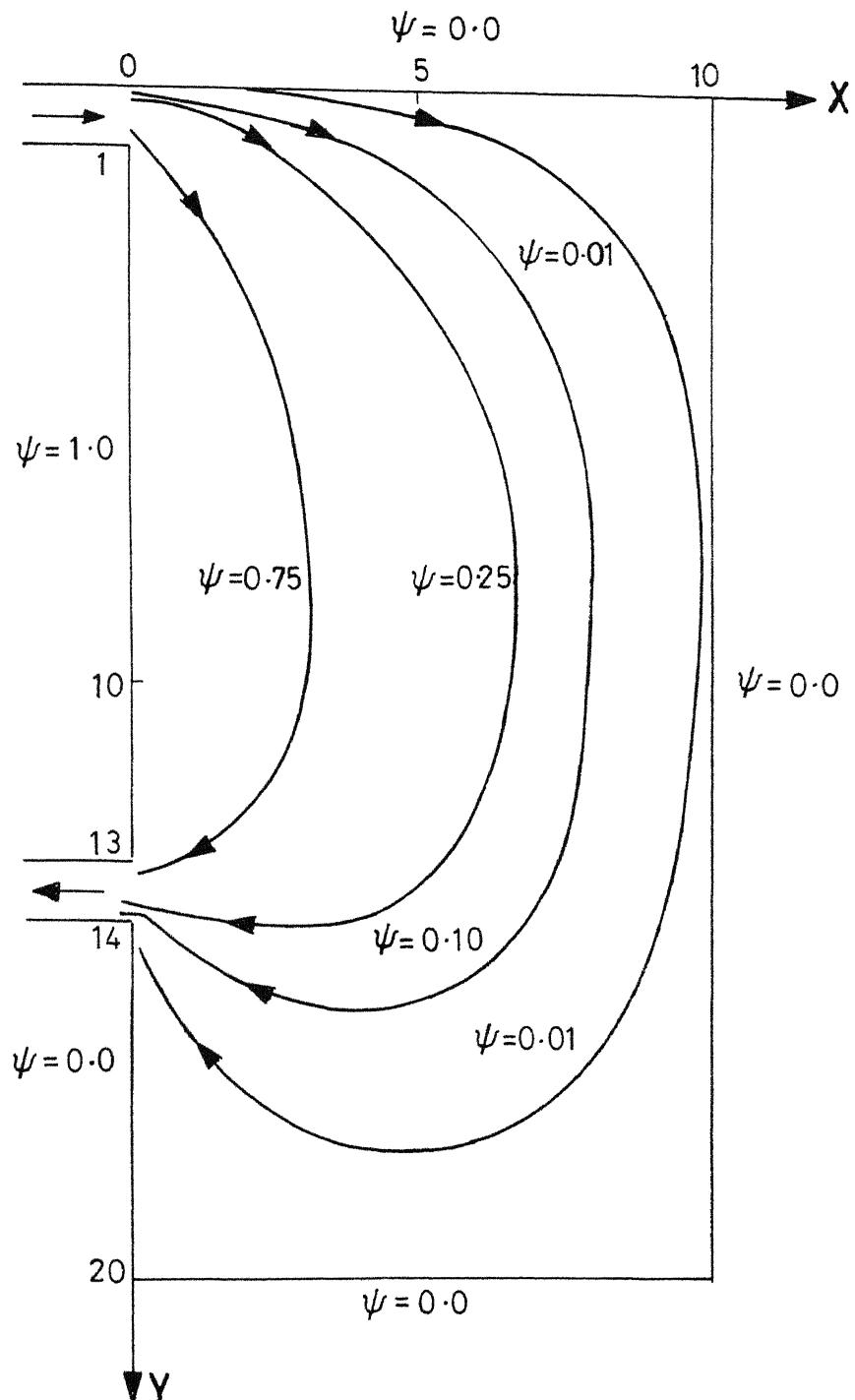


FIG. 5 EFFECT OF LOCATING THE OUTFALL, AWAY FROM THE EDGE OF THE WATER BODY, ON THE FLOW PATTERN IN THE TWO DIMENSIONAL HORIZONTAL CREEPING FLOW, WITH INTAKE AND OUTFALL ON THE SAME SIDE

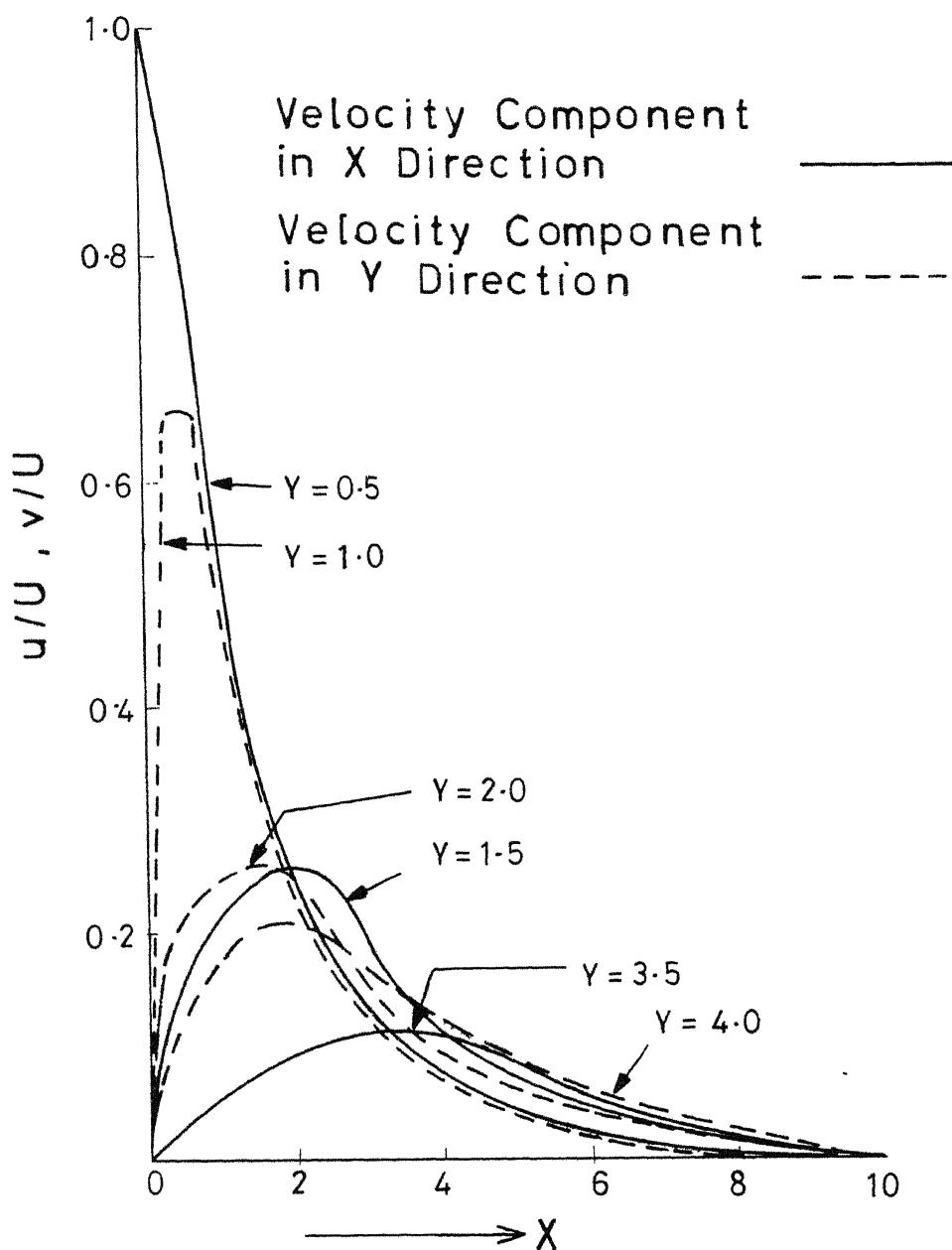


FIG. 6 VARIATION OF VELOCITY ALONG THE LONGITUDINAL DIRECTION FOR CREEPING FLOW, WITH INTAKE AND OUTFALL ON THE SAME SIDE

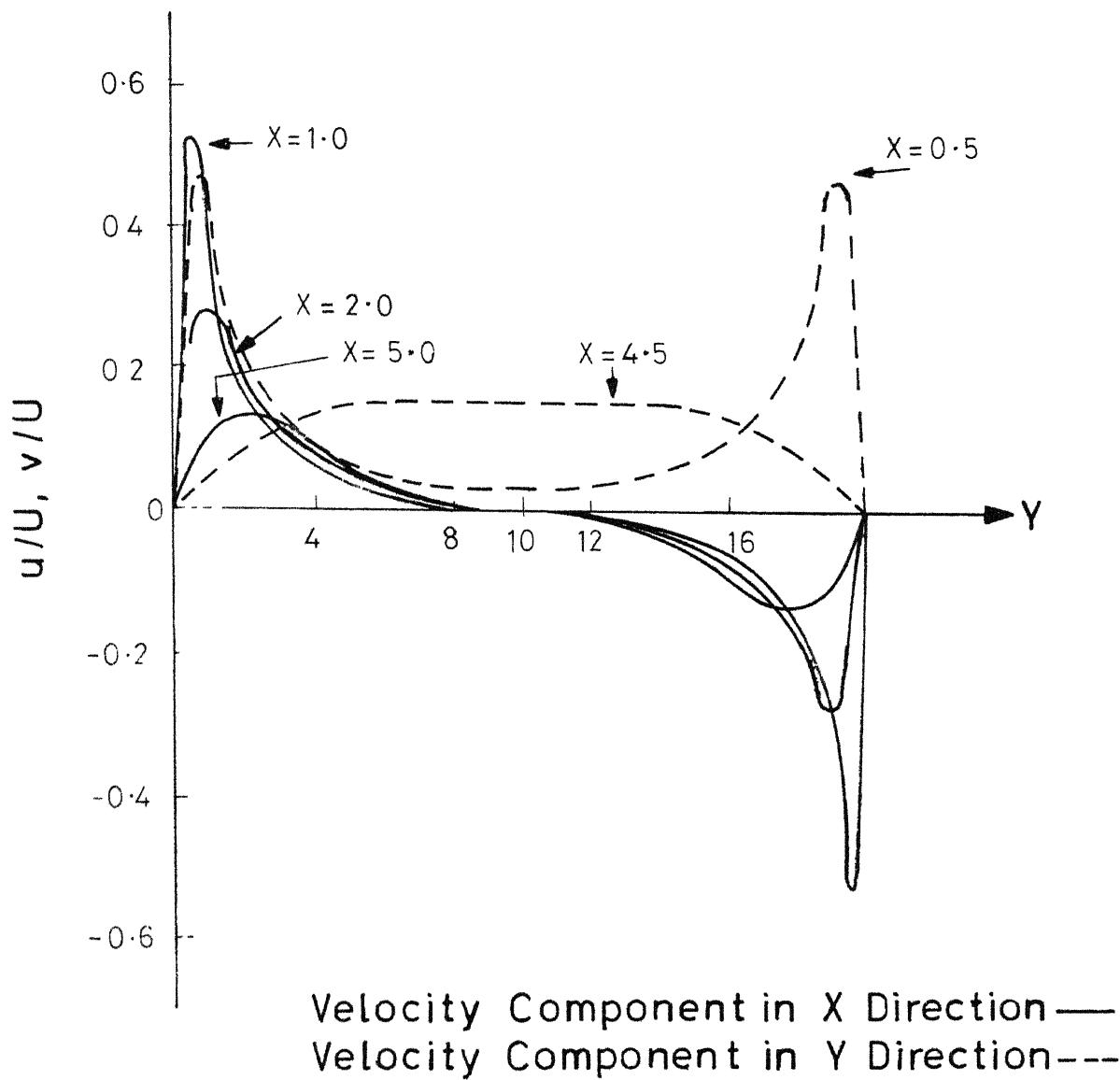


FIG. 7 VARIATION OF VELOCITY ALONG THE TRANSVERSE DIRECTION FOR CREEPING FLOW WITH INTAKE AND OUTFALL ON THE SAME SIDE

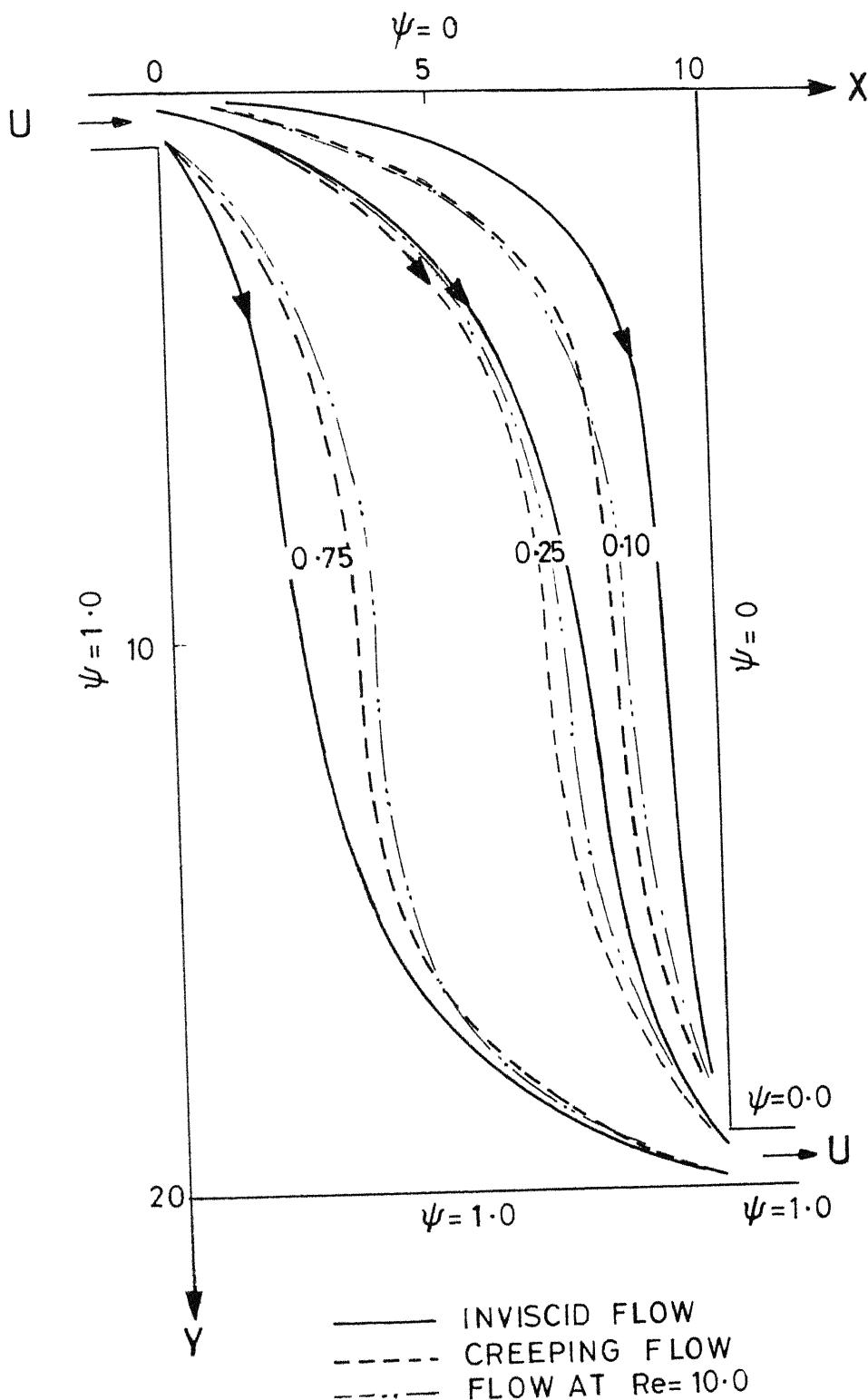


FIG. 8 FLOW PATTERN IN TWO DIMENSIONAL HORIZONTAL FLOW, WITH INTAKE AND OUTFALL ON THE OPPOSITE ENDS OF THE WATER BODY

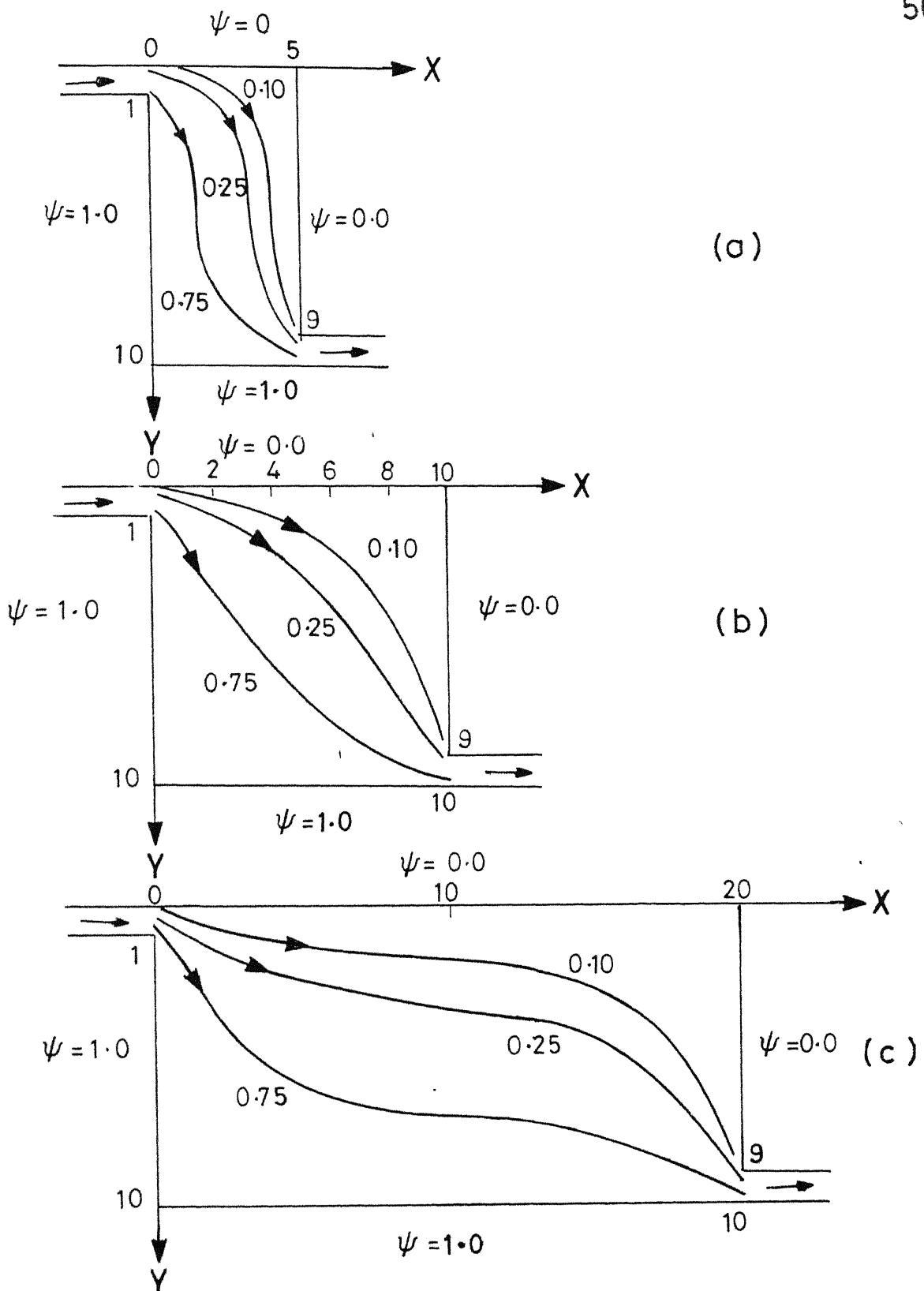


FIG.9 FLOW PATTERN IN THE TWO DIMENSIONAL HORIZONTAL CREEPING FLOW , WITH INTAKE AND OUTFALL ON THE OPPOSITE ENDS, FOR VARIOUS ASPECT RATIOS

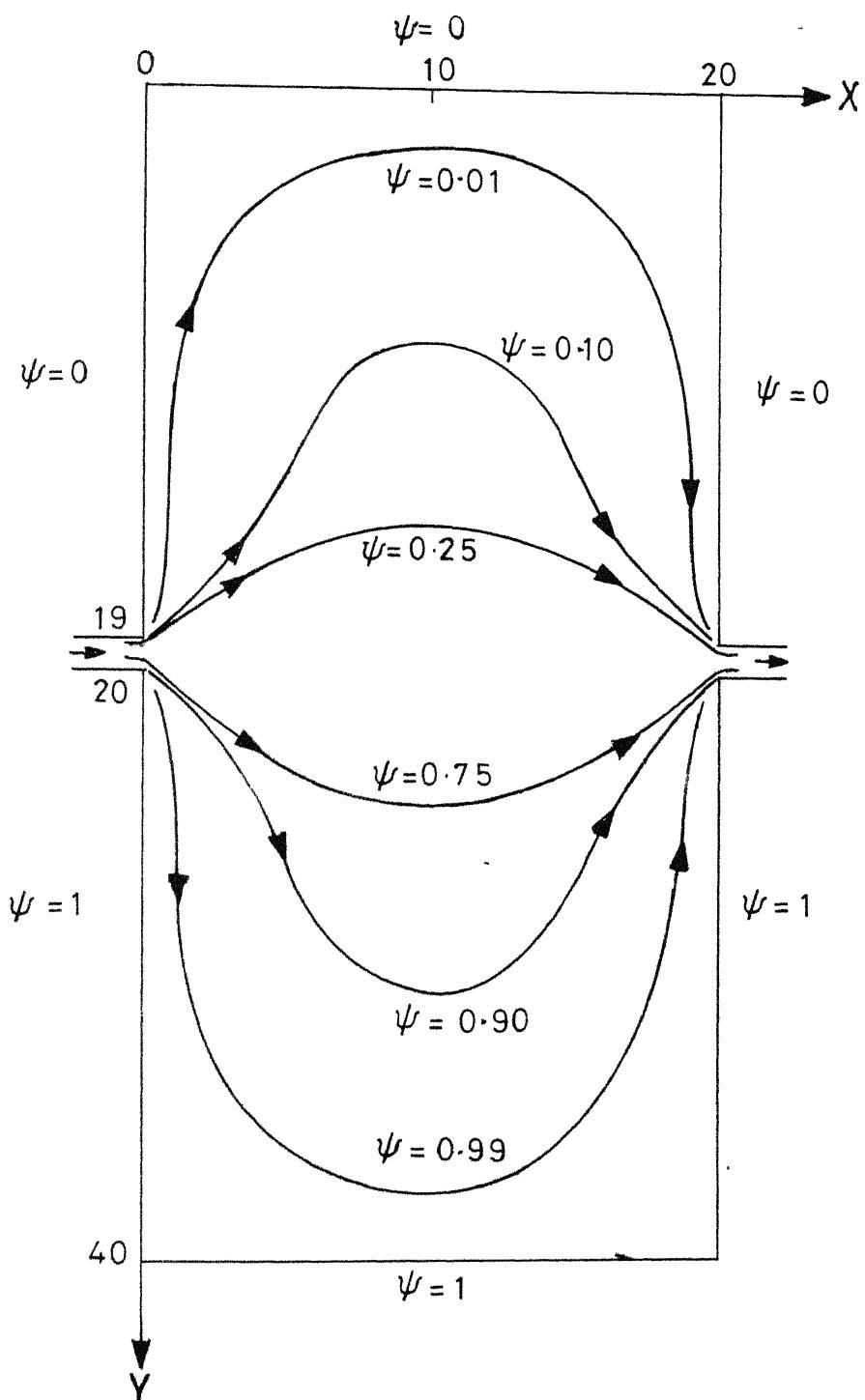


FIG.10 THE TWO DIMENSIONAL HORIZONTAL CREEPING FLOW FOR INTAKE AND OUTFALL LOCATED IN THE MIDDLE OF OPPOSITE SIDES OF THE WATER BODY

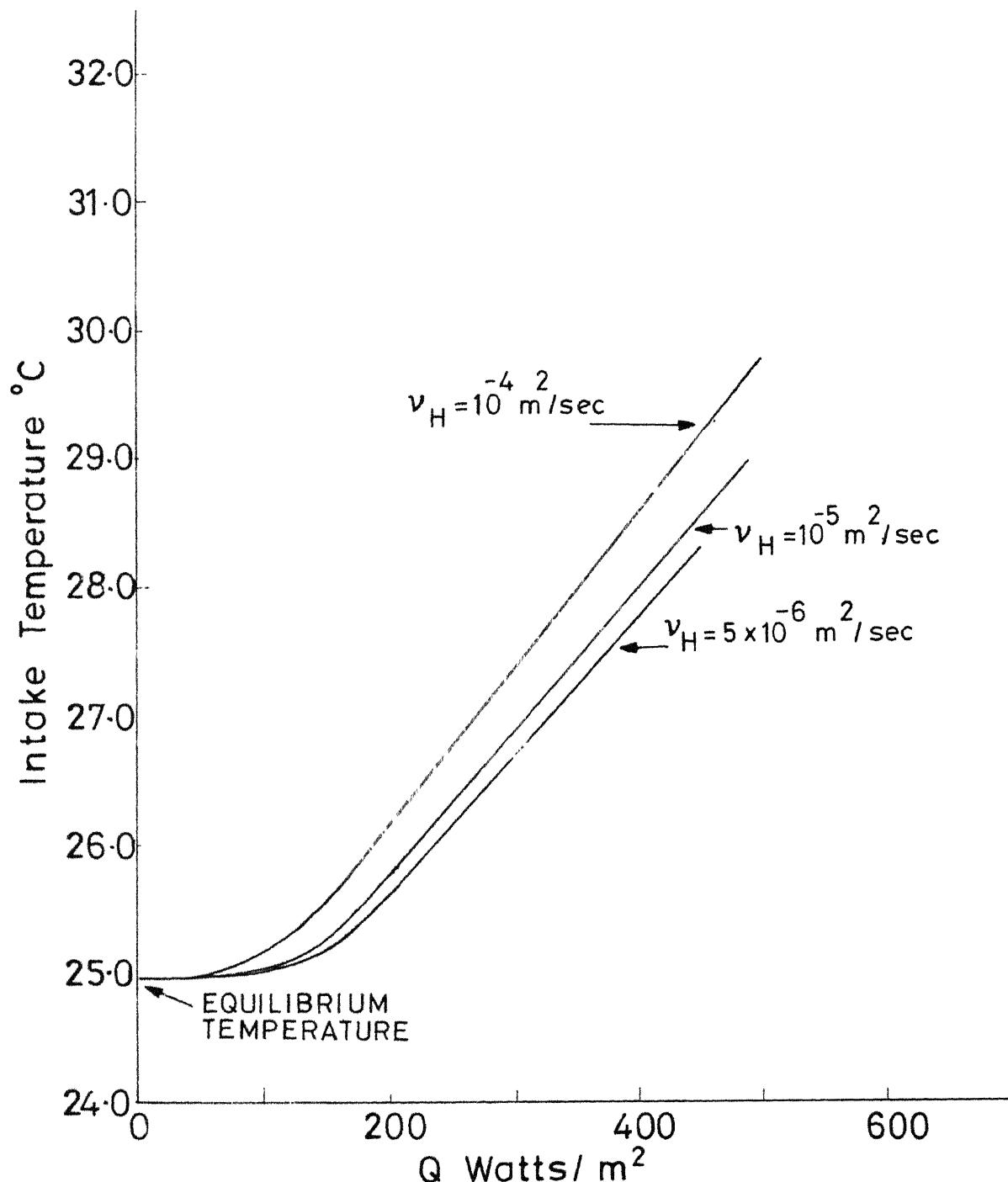


FIG.11 EFFECT OF POWER PLANT CAPACITY ON THE INTAKE TEMPERATURE FOR A ONE DIMENSIONAL HORIZONTAL FLOW MODEL

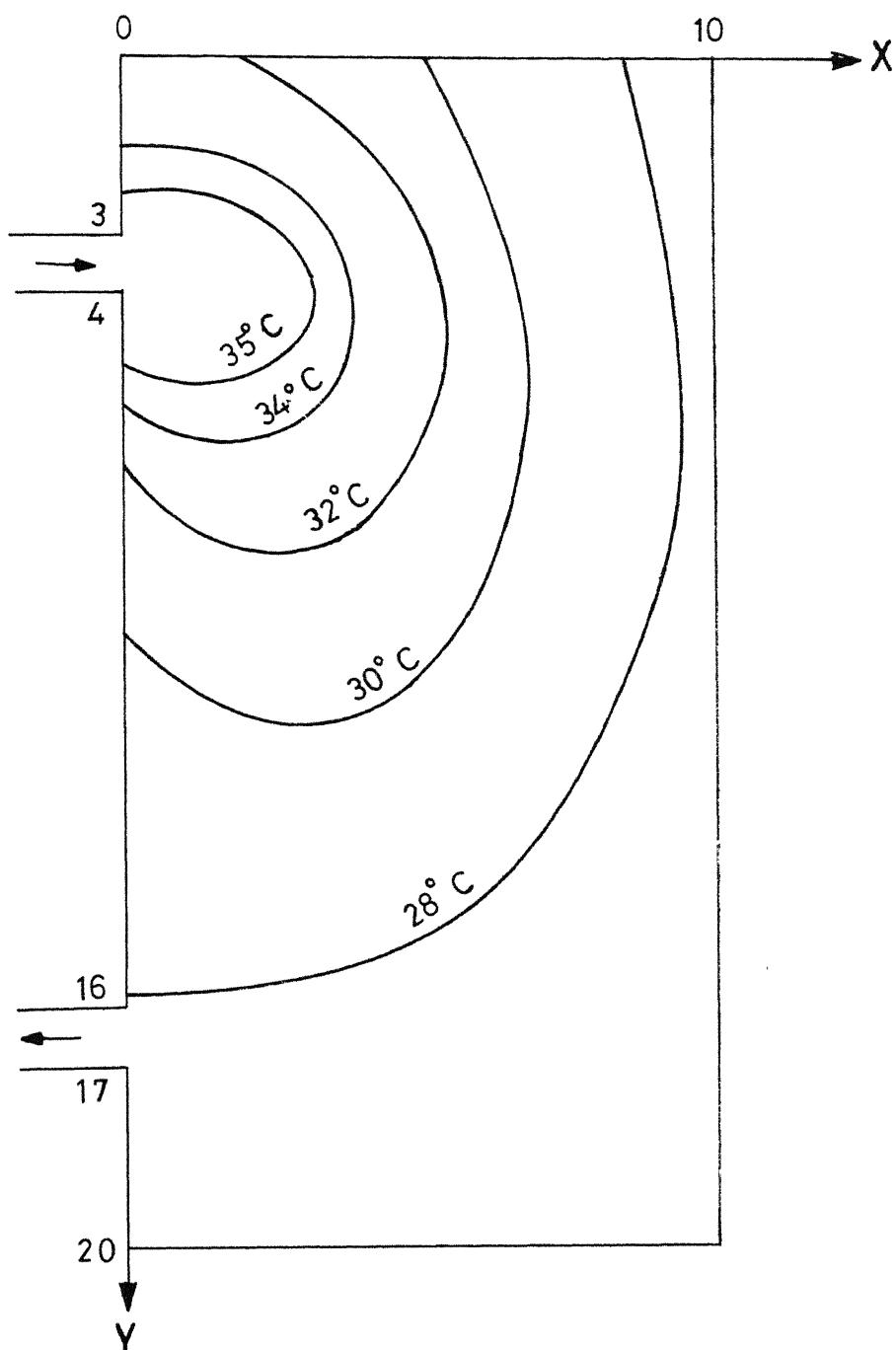


FIG. 14 ISOTHERMS IN THE TWO DIMENSIONAL  
HORIZONTAL CREEPING FLOW AT  
 $V_H = 2.5 \times 10^{-4} \text{ m}^2/\text{sec}$

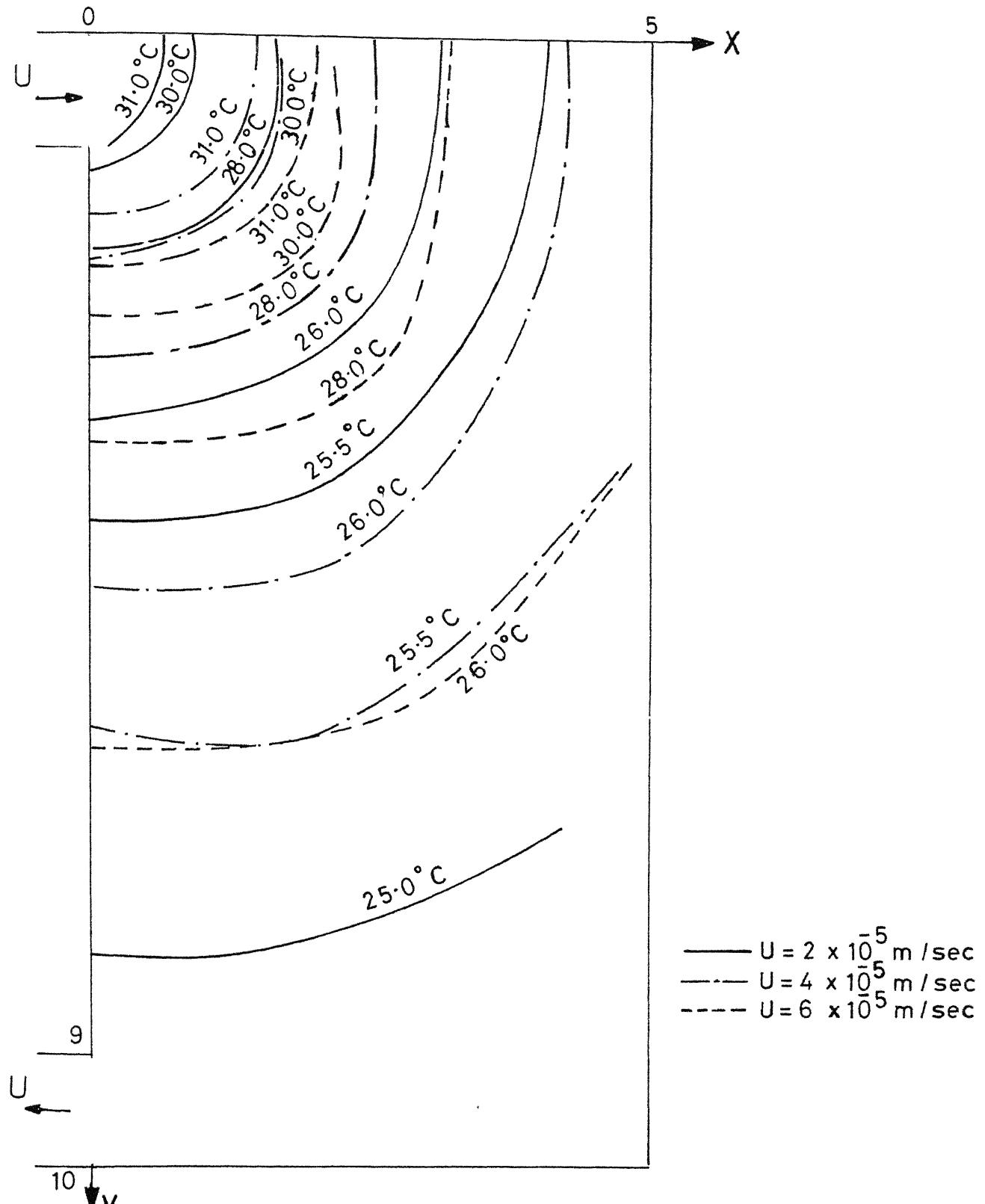


FIG.15 EFFECT OF A VARIATION IN OUTFALL VELOCITY  
ON THE ISOTHERMS IN THE TWO DIMENSIONAL  
HORIZONTAL CREEPING FLOW FOR  $\Delta T = 10^\circ\text{C}$

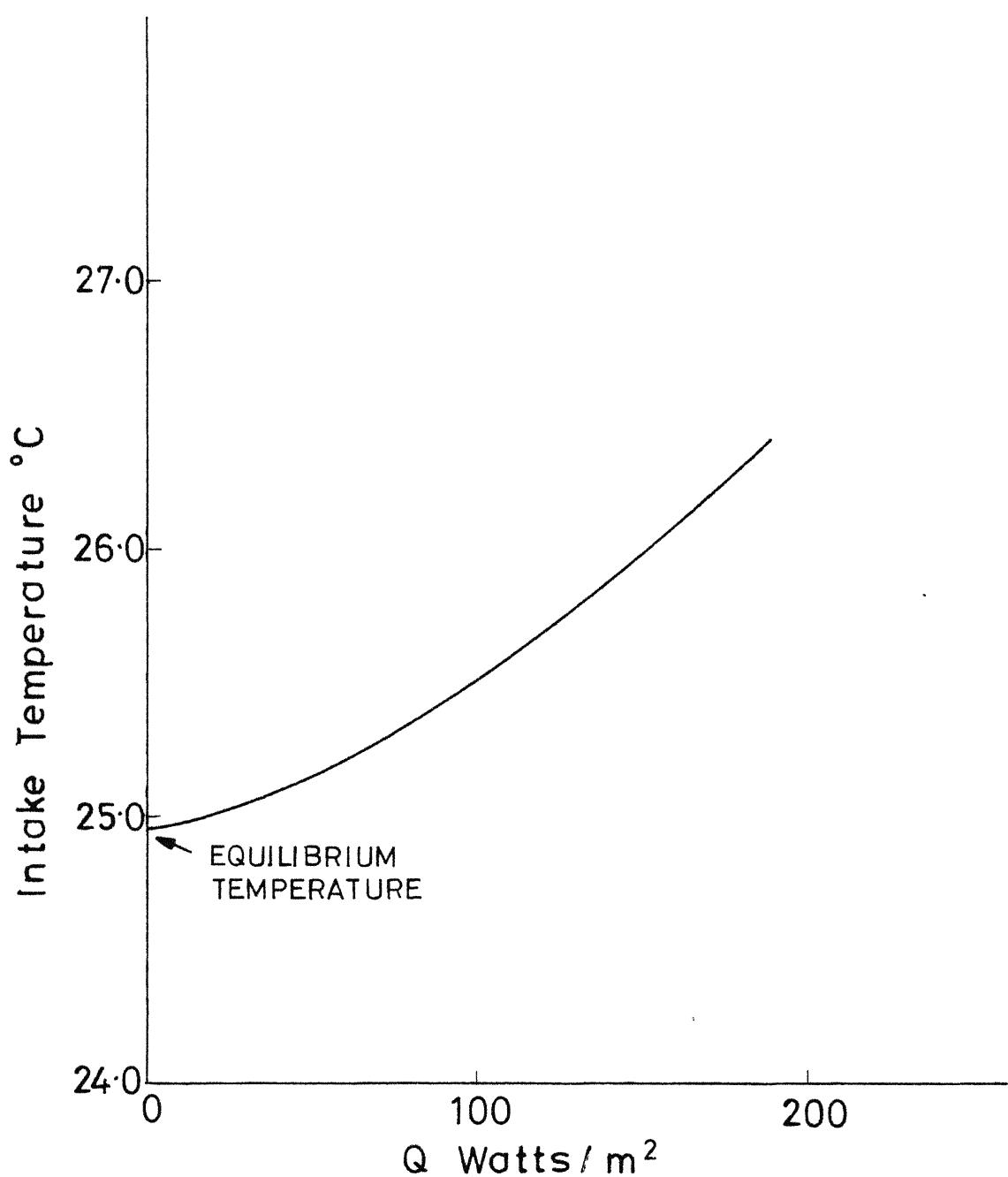


FIG.16 EFFECT OF POWER PLANT CAPACITY ON THE INTAKE TEMPERATURE IN THE TWO DIMENSIONAL HORIZONTAL CREEPING FLOW AT  $V_H = 2.5 \times 10^{-4} \text{ m}^2/\text{sec}$

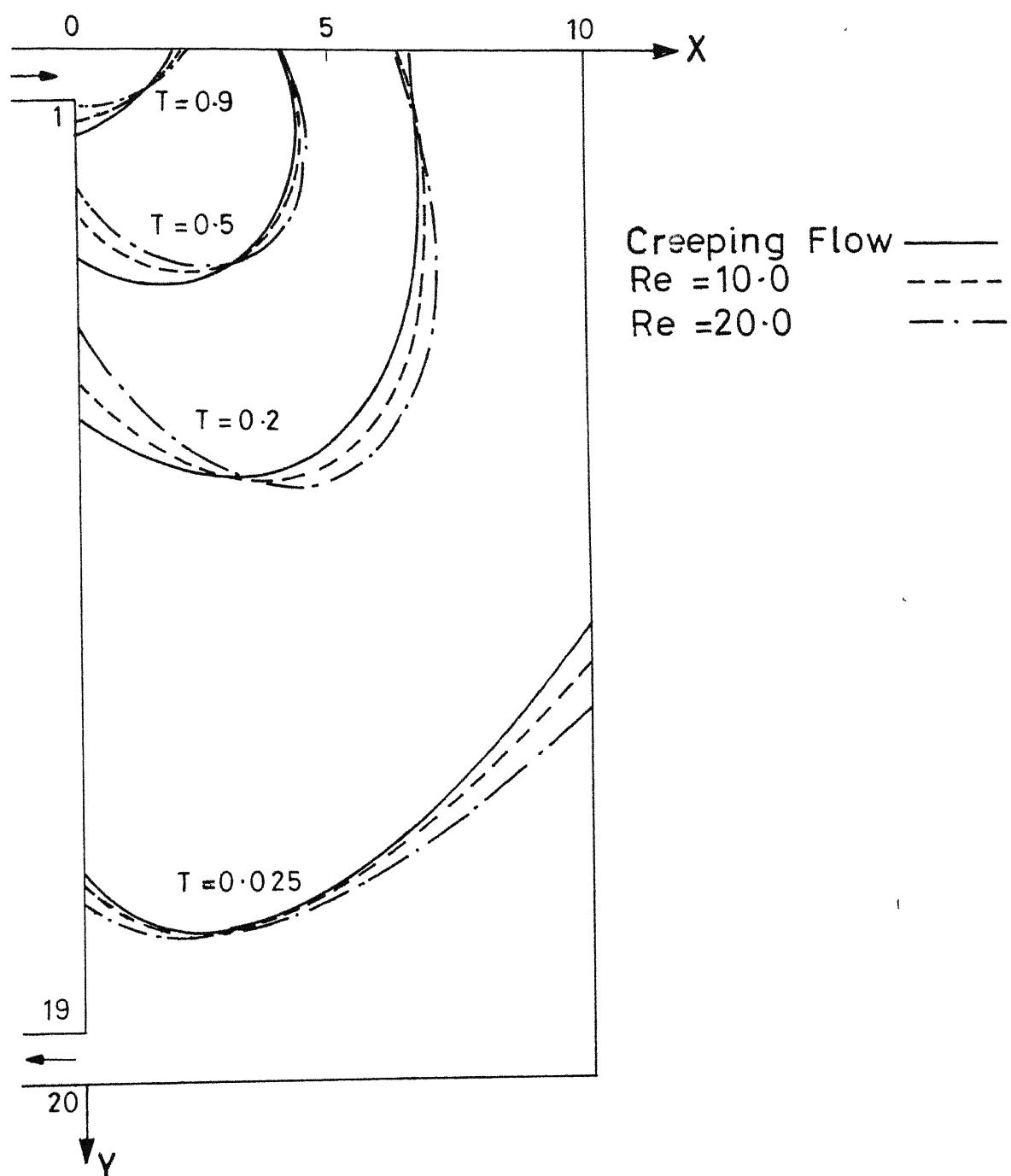


FIG.17 EFFECT OF REYNOLDS NUMBER ON THE ISOTHERMS IN THE TWO DIMENSIONAL HORIZONTAL FLOW AT  $\nu_H = 10^4 \text{ m}^2/\text{sec}$